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P-1408

TANK CONFERENCE

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TTMA

Truck Trailer Manufacturers Association

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December 5, 2000

Richard P. Bowling
President

Robert A. McGuire
Associate Administrator
Research & Special Programs Administration
United States Department of Transportation
400 Seventh Street, S.W.
Washington, DC 20590-0001

Subject: Incorporation by Reference of TTMA RP No. 96 in § 178.345-3

Dear Mr. McGuire:

Over the past decade a number of TTMA publications have been incorporated by reference in the hazardous materials regulations to reinforce requirements and assist in code compliance. Now, we propose your consideration of incorporating TTMA RP No. 96 in the structural integrity requirements for cargo tanks, § 178.345-3.

TTMA members have labored diligently over the past decade in drafting this document, which is intended to serve as a guide in performing structural integrity calculations for cargo tanks. We feel RP No. 96 accurately captures the letter and the intent of the HMR.

What's more this document has been used by Federal Motor Carrier Safety Administration (and former Federal Highway Administration) representatives in auditing cargo tank manufacturers.

Accordingly, we believe government's and industry's best interests are served by incorporating this RP by reference in § 178.345-3.

Sincerely,

Richard P. Bowling
President
dick@ttmanet.org

RPB:mm

Enclosure: TTMA RP No. 96

RP No. 96-97
December 1, 1997

Subject: **Structural Integrity of DOT 406, DOT 407, AND DOT 412 Cylindrical Cargo Tanks**
(Originally issued - May 31, 1996, and Revised December 1997)

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1.0 Preface:

1.1 Recommended Practices and Technical Bulletins are furnished by the TTMA as a guide to general practices in the manufacture, use, and repair of truck trailers. However, the scope of the TTMA's Recommended Practices and Technical Bulletins is not exhaustive of all general practices in the manufacture, use, and repair of truck trailers and there may exist such general practices which do not appear in either the Recommended Practices or Technical Bulletins.

1.2 Recommended Practices and Technical Bulletins represent the state-of-the-art that existed at the time of its preparation. Users of Recommended Practices and Technical Bulletins should familiarize themselves with advancements in practices that have occurred subsequent to the Recommended Practice's or Technical Bulletin's publication date.

1.3 The TTMA has not undertaken any evaluation of all the conceivable ways in which Recommended Practices or Technical Bulletins may be used by manufacturers, users, or repairers of truck trailers nor the consequences of such uses. Everyone who uses Recommended Practices or Technical Bulletins must first satisfy himself or herself that his or her safety, the safety of others, or the safety of the truck trailer and any other equipment will not be jeopardized by their use of information contained within the Recommended Practices or Technical Bulletins.

1.4 The Recommended Practices and Technical Bulletins may contain terms or words with specialized meanings. Definitions for such terms or words may be found in TTMA RP No. 36 - Tank Trailer and Tank Container Nomenclature or TTMA RP No. 66 - Trailer Nomenclature.

1.5 Within the Recommended Practices and Technical Bulletins, "shall" is used wherever conformance with the TTMA publication requires that there be no deviation from the specific recommendation. "Should" is used wherever deviation from the specific recommendation is permissible in complying with the TTMA publication.

1.6 Conformity with TTMA publications by manufacturers, users and repairers of truck trailers is voluntary and any non-conformity with such publications is not indicative of the non-conforming practice being deficient.

1.7 Any inclusion of Recommended Practices or Technical Bulletins within any contract, document or standard is voluntary, and any such inclusion shall not imply any endorsement or approval by the TTMA due to the multitude of ways in which the Recommended Practices or Technical Bulletins may conceivably be used.

2.0 Purpose:

2.1 To recommend acceptable methods of calculating the structural integrity of DOT 406, DOT 407, and DOT 412 cylindrical cargo tanks in conformance with 49 CFR 178.345-3 and 178.345-8(e).

3.0 Scope:

3.1 This Recommended Practice shows one method for calculating stresses in the cargo tank head and shell for compliance with the regulatory requirements. Other methods/equations may be equally acceptable.

49 CFR 178.345-3(a)(3) states that "Alternate test or analytical methods, or a combination thereof, may be used in place of the procedures described in paragraphs (b) [static loads], and (c) [static & dynamic loads] of this section, if the methods are accurate and verifiable." It is up to the Design Certifying Engineer to determine the applicability of this Recommended Practice to a cargo tank design. The Design Certifying Engineer shall perform any additional or other calculations to verify a particular cargo tank design.

3.2 Detailed calculations found in the ASME Code such as the determination of nozzle reinforcement, for example, are not included in this Recommended Practice, but shall be performed.

3.3 While this Recommended Practice pertains to both self-supporting tank trailers and cargo tanks mounted on truck chassis, it has primarily been developed for tank trailers. Some of the calculations may not be applicable, nor necessary for cargo tanks mounted on truck chassis. 178.345-3(e) states that -

> "For a cargo tank mounted on a frame or built with integral structural supports, the calculation of effective stresses for the loading conditions in paragraph (c) of this section [178.345-3(c)] may include the structural contribution of the frame or the integral structural supports."

3.4 This Recommended Practice describes procedures for determining the principal stresses in a cargo tank without consideration of stresses resulting from concentrated loads due to the attachment of cabinets and other readily attachable equipment.

Although stresses into the cargo tank due to overturn, rear-end, and bottom accident damage protection loads specified in 49 CFR 178.345-8 shall be considered, the means of including these loads in the structural integrity calculations are not described in this Recommended Practice.

Although the local stresses at saddles and bolsters due to the loads specified in 49 CFR 178.345-3 shall be considered, the means of calculating these stresses are not described in this Recommended Practice.

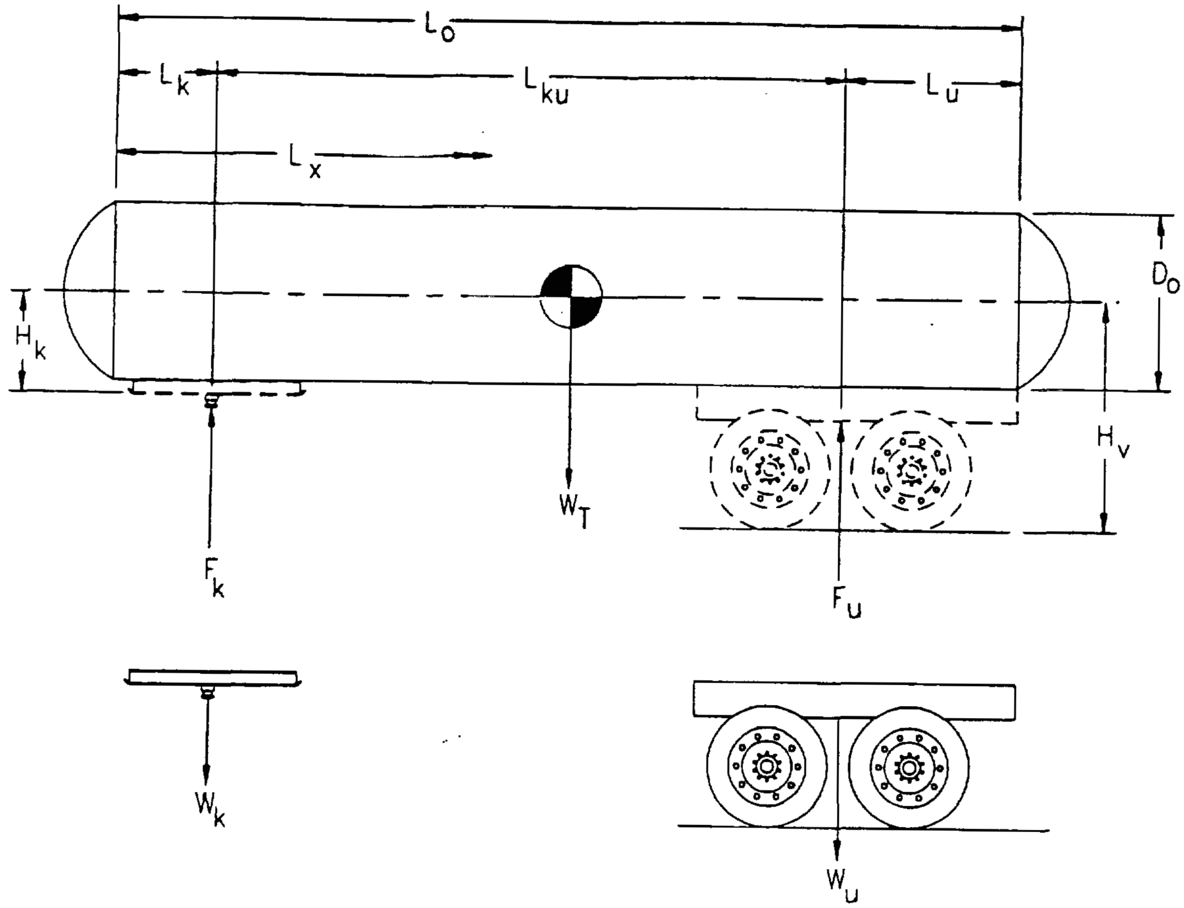
3.5 Additional calculations and considerations are required in designing a pressure vessel in Full conformance with the ASME Code.

3.6 Per 178.345-3(a)(2), the relevant physical properties of material used may be established by testing the material (or similar manufactured component parts) in conformance with a recognized national standard).

3.7 Rather than have separate calculations for all elements each time, general parameters for limiting cases may be established for various types of tanks and pressure ratings may be taken in proportion to allowable stresses when minimum thicknesses have been calculated.

3.8 Weld joint efficiency must be considered where applicable.

4.0 Definitions and Nomenclature:



A [dimensionless factor] = $\frac{0.125}{R_o/t_s}$

a [inches^2] = cross section area of shell

Note: For uniform thickness vessel shells only:

$$a = 2\Pi * \left[\frac{R_i + R_o}{2}\right] * ts$$

B [psi] = ASME allowable compressive stress due to static loads

b [dimensionless] = Number of baffles contributed to surge reduction (not to exceed 4)

c [inches] = distance from neutral axis to the most remote fiber of the section

D_i [inches] = inside diameter of shell

D_o [inches] = outside diameter of shell

e [dimensionless factor] [UG-27(b)] =
joint efficiency for, or the efficiency of, appropriate joint in cylindrical or spherical shells, or the efficiency of ligaments between openings, whichever is less.

For welded vessels, use the efficiency specified in UW-12 of the ASME Code.

E [psi] [UG-23(b)] = modulus of elasticity of material at design temperature

F_S = factor of safety for critical buckling stress

Fu [lbs] = reaction load of the cargo tank and lading at undercarriage

F_k [lbs] = reaction load of the cargo tank and lading at kingpin

H_k [inches] = height from the centerline of vessel to the horizontal pivot of the tractor or converter dolly fifth wheel, or the drawing hinge of the fixed dolly

H_v [inches] = height from road to centerline of vessel cross section

I [inch^4] = moment of inertia

J [inch^4] = polar moment of inertia

Note: For uniform thickness vessel shells only:

$$J = 2\Pi * \left[\frac{R_i + R_o}{2}\right]^3 * t_s$$

L_k [inches] = distance from cargo tank front head seam to center of kingpin

L_o [inches] = overall length of shell from front head seam to rear head seam

L_u [inches] = distance from cargo tank rear head seam to center of undercarriage, that is, halfway between the axles of a tandem, the second axle of a tridem, or the center of the axle of a single axle trailer. This symbol may not be applicable to spread axles or trailers with more than three axles at the rear.

L_{ku} [inches] = distance from centerline of kingpin to centerline of undercarriage

L_x [inches] = distance from cargo tank front head seam to any longitudinal location.

L_{max} [inches] = distance from cargo tank front head seam to point of maximum bending moment.

M [inch-lbs] = moment

P_d [psi] = dynamic impact pressure

P_h [psi] = static head pressure

P_L [psi] = dynamic and static pressure ($P_d + P_T$)

P_m [psi] = maximum gauge pressure permissible at the top of the cargo tank in its normal operating position at the operating temperature specified for that pressure.

P_T [psi] = total static pressure = $P_h + P_m$

P_v [psi] = external pressure rating

Q [dimensionless factor] [App. 1, par. 1-4(d) of the ASME Code] =
a factor in the formulas for torispherical heads depending on the head proportion R_h / R_k

R_h [inches] = inside radius of head (crown radius)

R_h/R_k [dimensionless factor] =
a ratio of the inside crown radius to the inside knuckle radius

R_i [inches] [UG-27(b)] =
inside radius of shell

R_o [inches] = outside radius of shell

R_k [inches] = knuckle radius

S [psi] [178.345-3(c)] =
effective stress, at any given point under combination of static and normal operating loadings, or static and an extreme dynamic loading

S_a [psi] = allowable static tensile stress for a particular material per Table UHA-23, UCS-23, or UNF-23 of the 1992 Edition, A93 Addenda of The ASME Code except as limited by footnote 1 of Appendix 1 of the ASME Code. [49 CFR 171.7(a), *Matter incorporated by reference-* recognizes only the 1992 Edition, A93 Addenda of the ASME Code and does not allow for current editions of the ASME Code to be incorporated.]

S_b [psi] = allowable compressive stress

S_{bA} [psi] = critical compressive buckling stress per the Alcoa formula

S_{bY} [psi] = critical compressive buckling stress per the Roark & Young formula

S_c [psi] = compressive stress due to static loads

S_d [psi] = effective tensile stress generated by a 2 "g" deceleration of liquid cargo

S_{dx} [psi] = longitudinal tensile stress generated by a 2 "g" deceleration of liquid cargo

S_{dy} [psi] = circumferential tensile stress generated by a 2 "g" deceleration of liquid cargo

S_m [psi] = allowable dynamic tensile strength for a particular material per 49 CFR 178.345-(3)(a)

S_u [psi] = ultimate tensile strength for a particular material

S_y [psi] [178.345-3(c)] =
circumferential stress generated by internal and external pressure, when applicable

S_x [psi] [178.345-3(c)] =
the net longitudinal stress, in psi, generated by the following loading conditions:

S_{x1},S_{x2},S_{x3},S_{x4},S_{x5},S_{x6},S_{x7},S_{x8}, and S_{x9}.

S_{x1} [psi] = The greater of longitudinal stresses resulting from the MAWP or from the lowest pressure at which the cargo tank may operate, in combination with the bending stress generated by the weight of the lading, the weight of the cargo tank and other structures and equipment supported by the cargo tank wall.

S_{x2} [psi] = The tensile or compressive stress generated by the axial load resulting from a longitudinal decelerative force equal to 0.7 times the vertical reaction at each suspension assembly, applied at the road surface. The vertical reaction must be calculated based on the static weight of the fully loaded cargo tank motor vehicle.

S_{x3} [psi] = The tensile or compressive stress generated by the bending moment resulting from a longitudinal decelerative force equal to 0.7 times the vertical reaction at each suspension assembly, applied at the road surface. The vertical reaction must be calculated based on the static weight of the fully loaded cargo tank motor vehicle.

S_{x4} [psi] = The tensile or compressive stress generated by the axial load resulting from a longitudinal accelerative force equal to 0.7 times the vertical reaction applied at the horizontal pivot of the upper coupler (fifth wheel) or turntable supporting the cargo tank motor vehicle.

S_{x5} [psi] = The tensile or compressive stress generated by the bending moment resulting from a longitudinal accelerative force equal to 0.7 times the vertical reaction, applied to the horizontal pivot of the upper coupler (fifth wheel) or turntable supporting the cargo tank motor vehicle.

S_{x6} [psi] = The tensile or compressive stress generated by the bending moment resulting from a vertically up accelerative force equal to 0.7 times the vertical reaction, applied at each suspension assembly. The vertical reaction must be calculated based on the static weight of the lading, the weight of the cargo tank and other structures and equipment supported by the cargo tank wall.

S_{x7} [psi] = The tensile or compressive stress generated by the axial load resulting from a longitudinal decelerative force equal to 0.7 times the vertical reaction applied at the horizontal pivot of the upper coupler (fifth wheel) or turntable supporting the cargo tank motor vehicle.

S_{x8} [psi] = The tensile or compressive stress generated by the bending moment resulting from a longitudinal decelerative force equal to 0.7 times the vertical reaction, applied to the horizontal pivot of the upper coupler (fifth wheel) or turntable supporting the cargo tank motor vehicle.

S_{x9} [psi] = The tensile or compressive stress generated by the bending moment resulting from a lateral accelerative force equal to 0.4 times the vertical reaction at each suspension assembly, applied at the road surface. The vertical reaction must be calculated based on the static weight of the fully loaded cargo tank motor vehicle.

S_s [psi] [178.345-3(c)] =

The net shear stress, in psi, generated by the following loading conditions:

S_{s1}, S_{s2}, and S_{s3}

S_{s1} [psi] The vertical shear stress generated by a vertical force equal to 1.0 times the vertical reaction, applied at each suspension assembly. The vertical reaction must be calculated based on the static weight of the lading, the weight of the cargo tank and other structures and equipment supported by the cargo tank wall.

S_{s2} [psi] The vertical shear stress generated by a vertical force equal to 0.7 times the vertical reaction, applied at each suspension assembly. The vertical reaction must be calculated based on the static weight of the lading, the weight of the cargo tank and other structures and equipment supported by the cargo tank wall.

S_{s3} [psi] = The lateral shear stress generated by a lateral accelerative force equal to 0.4 times the vertical reaction, applied laterally at the road surface. The vertical reaction must be calculated based on the static weight of the fully loaded cargo tank motor vehicle.

S_{s4} [psi] = The torsional shear stress generated by a lateral accelerative force equal to 0.4 times the vertical reaction, applied laterally at the road surface. The vertical reaction must be calculated based on the static weight of the fully loaded cargo tank motor vehicle.

t [inches] = ASME calculated minimum thicknesses of heads and shell

t_h [inches] = Minimum thickness of head less corrosion allowance

t_s [inches] = Minimum thickness of shell less corrosion allowance
Note:
The minimum thickness for the shell and heads less corrosions allowance must be such that the maximum stress levels specified in 178.345-3 are not exceeded. In no case may the shell or head thickness be less than that specified in the applicable specification.

T [inch-lbs] = torque

V [gallons] = volume of tank

v [dimensionless] = Poisson's ration

W_b [lbs/gal] = maximum density of product which can be carried in a cargo tank which may not be fully loaded

W_c [lbs/gal] = maximum density of cargo tank contents for a fully loaded cargo tank

W_k [lbs] = weight of kingpin/upper coupler

W_L [lbs] = maximum weight of product which can be carried in the vessel

W_s [lbs] = weight of empty vessel = tare weight - W_k - W_u

W_T [lbs] = weight of vessel and contents = $W_s + W_L$

W_{Ti} [lbs/inch] = weight of vessel and contents per unit length

W_u [lbs] = weight of the undercarriage

Z_v [inch^3] = section modulus of the vessel = I / c

Note: For uniform thickness vessel shells only

$$Z_v = \Pi * \left[\frac{R_i + R_o}{2}\right]^2 * t_s$$

5.0 General Assumptions:

5.1 The following assumptions have been made by TTMA.

5.1.1 The weight of manhole assemblies, baffles, valves, piping, heads, and other attachments to the cargo tank vessel are assumed to be uniformly distributed over the length of the vessel. The weight of the upper coupler and undercarriage subframe may also be included in this uniformly distributed weight.

5.1.2 The cargo tank motor vehicle does not experience longitudinal acceleration and deceleration simultaneously.

5.1.3 The cargo tank motor vehicle does not experience full vacuum while loaded.

5.1.4 When determining the allowable strength for dynamic loads, combined static and dynamic loads, and accident induced loads, a weld joint efficiency should be considered. The Design Certifying Engineer should determine an appropriate value for joint efficiency based on ASME, UW-12, testing, or other appropriate means. The weld joint efficiency as found in UW-12 of the ASME Code need not be applied to these load cases per Docket No. HM-183C; Notice No. 93-7, as published in the Federal Register, Vol. 58, No. 40, March 3, 1993, page 12319, which states:

> "... these efficiencies are not applicable to analyses of dynamic loads and combined static and dynamic loads such as those described in paragraph (c)" of 178.345-3.

5.1.5 In calculations involving head or shell radius or diameter as a factor in the formula, manufactured dimensions may be used. Calculations which involve thickness as a factor in the formula must use the minimum thickness after corrosion and not the manufactured thickness.

6.0 Calculation of Head and Shell Thickness and Stresses Based on Static Loads:

6.1 According to 178.345-3(b), the static design and construction of each cargo tank must be in accordance with Section VIII, Division 1 of the ASME Code. The tank design must include calculation of stresses generated by the MAWP, the weight of the lading, the weight of structures supported by the cargo tank wall and the effect of temperature gradients resulting from lading and ambient temperature extremes. When dissimilar materials are used, their thermal coefficients must be used in the calculation of thermal stress.

Note: 49 CFR 171.7(a), *Matter incorporated by reference*- recognizes only the 1992 Edition, A93 Addenda of the ASME Code and does not allow for current editions of the ASME Code to be incorporated.

(1) Stress concentrations in tension, bending and torsion which occur at pads, cradles, or other supports must be considered in accordance with Appendix G of Section VIII, Division 1 of the ASME Code.

(2) Longitudinal compressive buckling stress for ASME certified vessels must be calculated using paragraph UG-23(b), Section VIII, Division 1 of the ASME Code. For cargo tanks not required to be certified in accordance with the ASME Code, compressive buckling stress may be calculated using alternative analysis methods which are accurate and verifiable. When alternative methods are used, calculations must include both the static loads described in this paragraph and the dynamic loads described in 178.345-3(c).

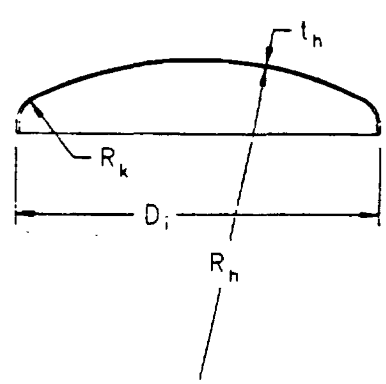
(3) 178.345-3(a)(4) states that "Corrosion allowance material may not be included to satisfy any of the design calculation requirements of this section."

6.2 Torispherical Head Thickness

The required thickness of a torispherical head per the ASME Code, Appendix 1, paragraph 1-4(d) is -

$$t = \frac{P_T R_h Q}{2S_a e - 0.2P_T}$$

Where: $Q = 1/4 \left(3 + \sqrt{\frac{R_h}{R_k}}\right)$



Torispherical heads made of materials having a specified minimum tensile strength per ASME exceeding 80,000 psi shall be designed using a value of S equal to 20,000 psi at room temperature and reduced in proportion to the reduction in maximum allowable stress values at temperature for the material as shown in the appropriate ASME Code Table of Subsection C, Requirements Pertaining to Classes of Materials.

The required thickness of a torispherical head for the case in which the knuckle radius is 6% of the inside crown radius and the inside crown radius equals the outside diameter of the skirt, shall be determined by

$$t = \frac{0.885\, P_T R_h}{S_a e - 0.1\, P_T}$$

per UG-32(e) of the ASME Code

where:

$$P_T = P_m + P_h$$

$$P_h = (.4336) * \left[\frac{D_l}{12}\right]\left[\frac{W_c}{8.3}\right]$$

where:

.4336 is the conversion from feet of water to psi

$\frac{D_l}{12}$ = the height of the vertical static head

$\frac{W_c}{8.3}$ = the specific gravity of the lading

Note: For non-ASME cargo tanks with flanged and dished heads, this equation may be used without the knuckle radius and crown radius limitations of ASME UG-32.

6.3 Determination of Shell Thickness Based on Longitudinal Tensile Stress (Circumferential Joints) for circular tanks.

According to UG-27(c) of the ASME Code, when the thickness does not exceed one-half of the inside radius, or P_T does not exceed 1.25 Se, the following formulas shall apply:

$$t' = \frac{P_T \; R_i}{2\, S_a e + 0.4\, P_T}$$

The formula will govern only when the circumferential joint efficiency is less than one-half the longitudinal joint efficiency, or when the effect of supplementary loadings (UG-22 of the ASME Code) causing longitudinal bending or tension in conjunction with internal pressure is being investigated.

Determination of minimum thickness for static bending stress for circular tanks follows:

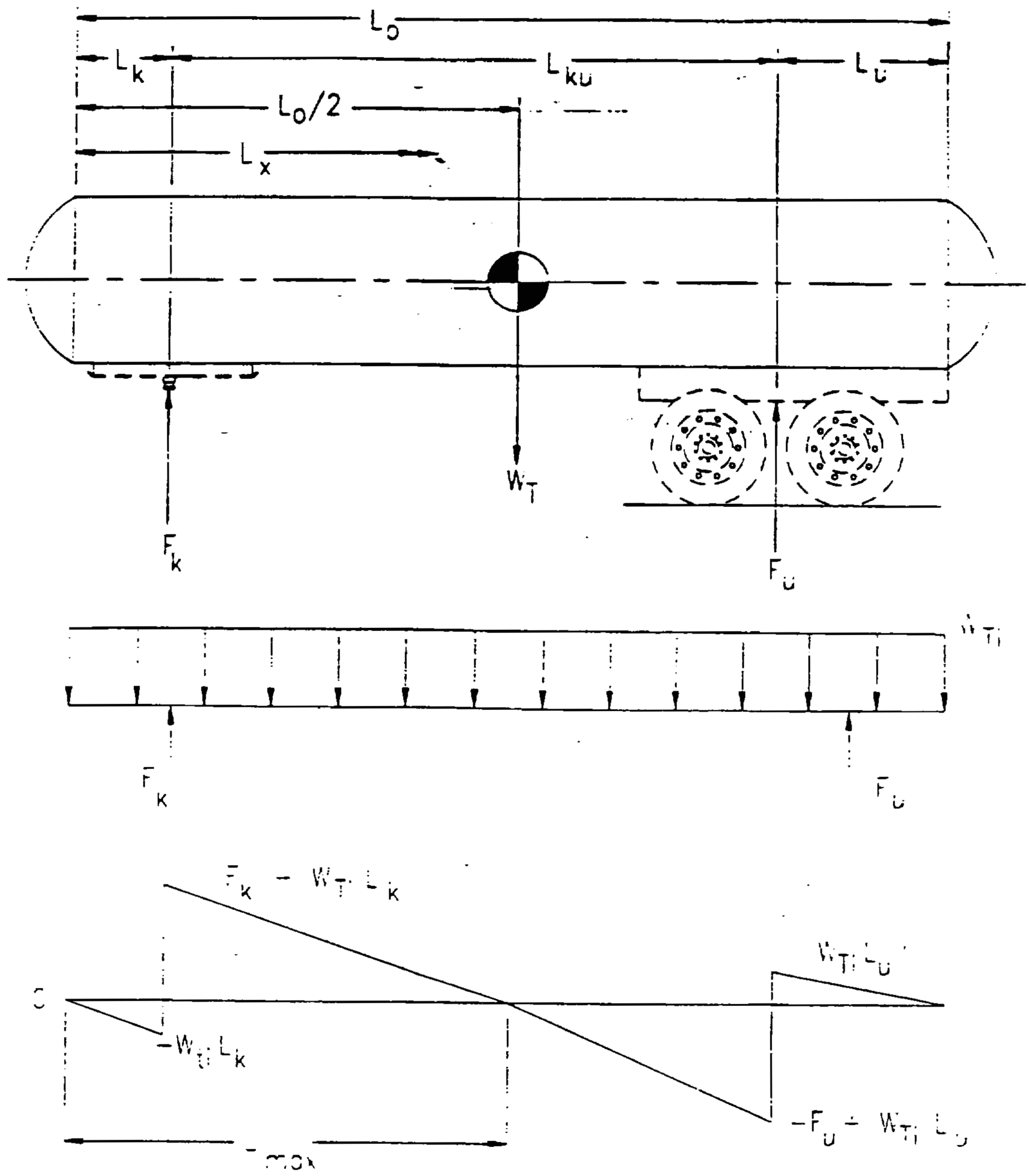
Static bending stress must be considered per UG-22 of the ASME Code. The formula below is found in Appendix L of the ASME Code.

$$t'' = M_{max} \qquad \text{(for uniform thickness vessel shells only)}$$

$$= \Pi \left[\frac{R_i + R_o}{2} \right]^2 S_a e$$

Determination of minimum thickness for total longitudinal stress for circular tanks is given by:

$$t = t' + t''$$



$W_{Ti} = W_T/L_o$

In equilibrium, the sum of the movements about any point = 0 Therefore:

$$\overset{+}{\curvearrowright} \quad \Sigma M_u = 0$$

$$-(W_{Ti})(L_o)(L_o/2-L_u) + (F_k)(L_{ku}) = 0$$

$$F_k = \frac{(W_{Ti})(L_o)(L_o/2-L_u)}{L_{ku}}$$

In equilibrium, the sum of all vertical forces = 0. Therefore:

$$+ \sum F_y = 0$$

$$-(W_{Ti})(L_o) + F_k + F_u = 0$$

$$F_u = (W_{Ti})(L_o) - F_k$$

The moment at any longitudinal location, L_x is:

$$M_x = \frac{-(W_{Ti})(L_x^2)}{2} + F_k (L_x - L_k) + F_u \quad (L_x - (L_{ku} + L_k))$$

for $L_x > (L_{ku} + L_k)$

$$M_x = \frac{-(W_{Ti})(L_x^2)}{2} + F_k (L_x - L_k)$$

for $L_k < L_x < (L_k + L_{ku})$

$$M_x = \frac{-(W_{Ti})(L_x^2)}{2}$$

for $L_x < L_k$

The location of the maximum moment is where the derivative of the moment with respect to distance is zero.

$$\frac{dM_x}{dL_x} = 0$$

This condition occurs only between F_k and F_u.

Therefore, use M_x equation for $L_k < L_x < (L_k + L_{ku})$

$$M_x = \frac{-(W_{Ti} L_x^2)}{2} + F_k (L_k - L_k)$$

$$\frac{dM_x}{dL_x} = (-W_{Ti})(L_x + F_k = 0$$

$$L_x = L_{max} = F_k/W_{Ti}$$

6.4 Determination of Shell Thickness Based on Circumferential Stress (Longitudinal Joints)

According to UG-27(c) of the ASME Code, when the thickness does not exceed one-half of the inside radius, or P_T does not exceed 0.385 $S_a e$, the following formulas shall apply:

$$t = \frac{P_T R_i}{S_a e - 0.6 P_T}$$

6.5 Longitudinal Compressive Stress

ASME Tanks:

If the cargo tank is certified in conformance to the ASME Code, then the allowable longitudinal compressive stress shall be calculated per UG-23(b) of the ASME Code.

Note: UG-23(b) may be the controlling equation of shell thickness for small diameter ASME tanks.

Non-ASME Tanks:

The allowable compressive stress for cargo tanks which are not required to be constructed and certified in conformance with the ASME Code may be determined by either of the following formulas and need not comply with paragraph UG-23(b) of the ASME Code. However, in no case shall the calculated compressive stress be greater than 25 percent of the ultimate tensile strength of the material.

If the top surface of the shell of the cargo tank is continuously reinforced, for instance by full length overturn protection, the longitudinal compressive stress in the reinforcement and the shell may be calculated based on the reinforcement and the shell acting as an integral beam and shall not exceed 25 percent of the ultimate tensile strength of the respective materials.

Also, the calculated compressive stress in the shell may not exceed the allowable compressive stress in the shell as calculated by the following Alcoa & Roark & Young formulas.

To consider dynamic loads, multiply the static bending moment by a factor of 1.7 when determining longitudinal compressive stress.

Also consider static bending plus the greatest external pressure applicable.

Also consider the longitudinal compressive stress resulting from 1.35g vertical acceleration plus 0.35g longitudinal deceleration at the trailer suspension.

For the following two formulas of allowable compressive stress, the Design Certifying Engineer should determine the factor of Safety (FS); Suggested FS = 1.5.

Formulas based on that found in the, *ALCOA STRUCTURAL HANDBOOK*, 1960, page 156 and Table 23.

Allowable compressive, $S_c = \dfrac{S_{bA}}{FS} = \dfrac{(\Pi/4)^2 E}{FS \left[\dfrac{R_i}{t_s}\right] \left[1 + \dfrac{\left[\dfrac{R_i}{t_s}\right]^{1/2}}{35}\right]^2}$

where:

R_i/t_s is greater than 200.

Formula from *FORMULAS FOR STRESS AND STRAIN*, Fifth Edition, by Roark and Young, pages 554 and 555, Table 35, Item 15.

Allowable compressive buckling stress $S_c = \dfrac{S_{bY}}{FS} = \dfrac{E\, t_s}{(FS)\, 2(3)^{1/2} [1-v^2]^{1/2} R_i}$

where:

R_i/t_s is greater than 10

[Based on the range of cargo tank radii and shell thicknesses, R_i/T_s is restricted to values greater than 200 and less than 400.]

L is greater than $17.2\,(R_i t_s)^{1/2}$

6.6 Determination of S_y Circumferential stress Due to Pressure

$$S_y = \frac{(P_m)\ (R_i)}{t_s} \qquad \text{at top centerline of tank}$$

$$S_y = \frac{(P_m + P_h/2)\ R_i}{t_s} \qquad \text{at side centerline of tank}$$

$$S_y = \frac{(P_T)\ (R_i)}{t_s} \qquad \text{at bottom centerline of tank}$$

6.7 Determination of S_{x1} Longitudinal Stresses Resulting from the MAWP in Combination with Static Bending Stress

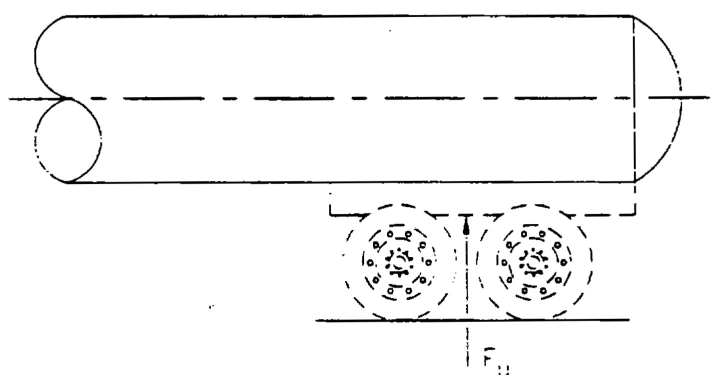
$$S_{x1} = \frac{(P_m)\ (R_i)}{2\ t_s} - \frac{M_{max}}{Z_v} \qquad \text{at top centerline of tank}$$

$$S_{x1} = \frac{[P_m + P_h/{}^2]\ (R_i)}{2\ t_s} \qquad \text{at side centerline of tank}$$

$$S_{x1} = \frac{(P_T)\ (R_i)}{2\ t_s} + \frac{M_{max}}{Z_v} \qquad \text{at bottom centerline of tank}$$

Note: S_{x1} shall also be calculated with the lesser of $P_m = 0$ or $P_m = P_v$

6.8 Determination of S_{s1} vertical shear stress generated by static weight:



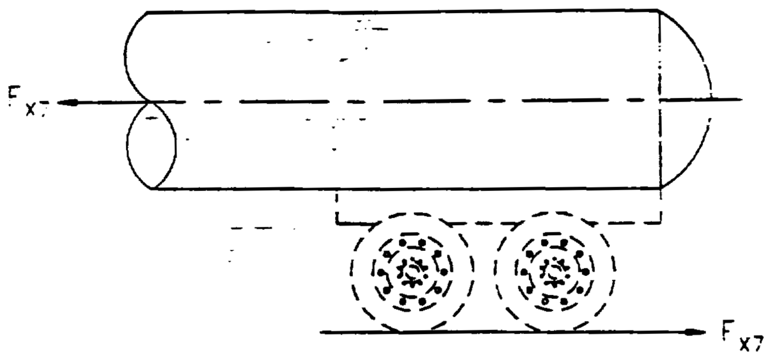
$$S_{s1} = \frac{F_u}{a/2} \qquad \text{at side centerline of tank}$$

$$S_{s1} = 0 \qquad \text{at top and bottom centerline of tank}$$

Refer to shear diagram on page 17 for a more accurate calculation of shear force along the length of the trailer.

7.0 Calculation of Head and Shell Stresses Due to Dynamic Loads:

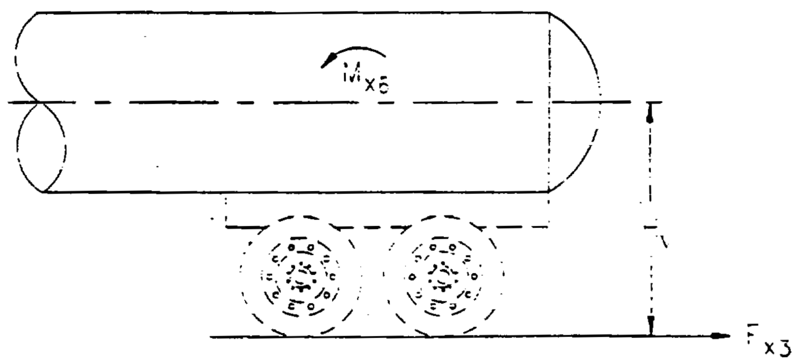
7.1 **Determination of S_{x2} the stress generated by the axial load resulting from a longitudinal decelerative force at the trailer suspension.**



$S_{x2} = \frac{F_{x2}}{a}$ At Top, Side & Bottom Centerline of Tank

where: $F_{x2} = 0.70 \; (F_u + W_u)$

7.2 Determination of S_{x3}, the stress generated by the bending moment resulting from a longitudinal decelerative force at the trailer suspension.



$S_{x3} = \frac{-M_{x3}}{Z_v}$ at top centerline of tank

$S_{x3} = 0$ at side centerline of tank

$S_{x3} = \frac{M_{x3}}{Z_v}$ at bottom centerline of tank

where $M_{x3} = F_{x3} \, H_v$

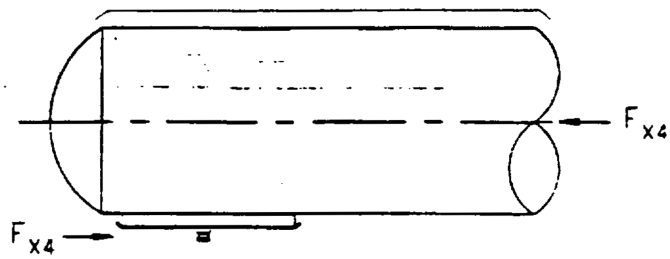
Note: This equation gives the maximum value of M_{x3}. Upon analysis by a Design Certifying Engineer, M_{x3} may be found to decrease of the length of a cargo tank motor vehicle depending upon support assumptions.

but $F_{x3} = F_{x2}$

then $S_{x3} = \frac{0.70 \; (F_u + W_u) \; H_v}{Z_v}$

7.3 Determination of S_{x4}, the tensile stress generated by the axial load resulting from a longitudinal accelerative force.

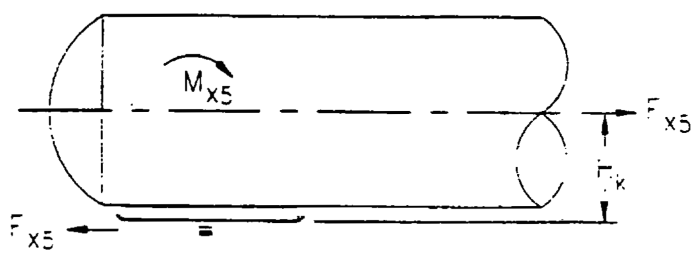
7.4



$S_{x4} = \frac{F_{x4}}{a}$ at top, side, & bottom centerline of tank

where $F_{x4} = 0.70\ (F_k + W_k)$

7.4 **Determination of S_{x5} stress generated by the bending moment resulting from a longitudinal accelerative force.**



$S_{x5} = \frac{-M_{x5}}{Z_v}$ at top centerline of tank

$S_{x5} = 0$ at side centerline of tank

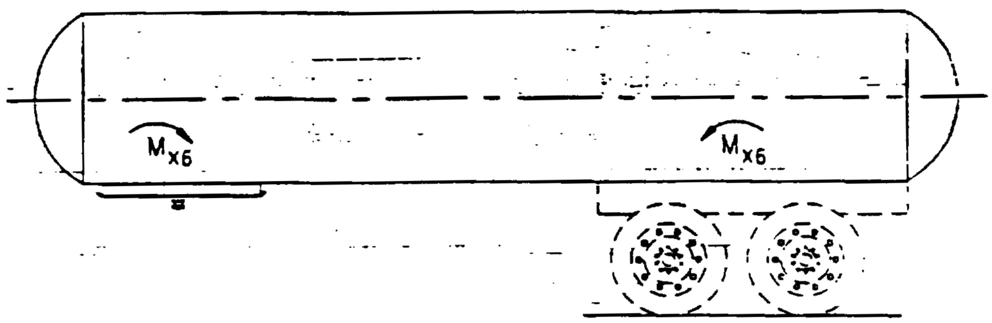
$S_{x5} = \frac{M_{x5}}{Z_v}$ at bottom centerline of tank

where $M_{x5} = F_{x5} H_k$

Note: This equation gives the maximum value of M_{x5}. Upon analysis of a Design Certifying Engineer, M_{x5} may be found to decrease over the length of a cargo tank motor vehicle depending upon support assumptions.

but $F_{x5} = F_{x4} = 0.70\ (F_k + W_k)$

7.5 Determination of S_{x6} stress generated by a bending moment resulting from an upward accelerative force.



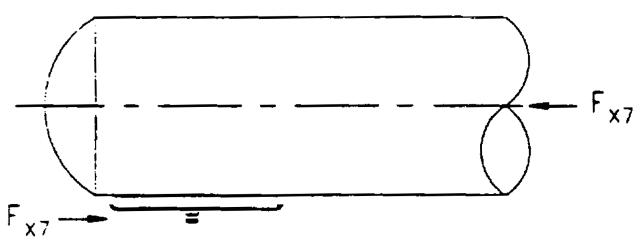
The reason the factor 0.7 is used instead of 1.7 is that this stress is added in the determination of S_{xc} with S_{x1} which includes the factor 1.0.

$S_{x6} = \frac{(0.7)\ M_{x6}}{Z_v} = \frac{(0.7)\ M_{max}}{Z_v}$ **at bottom centerline of tank**

$S_{x6} = 0$ **at side centerline of tank**

$S_{x6} = \frac{(0.7)\ M_{max}}{Z_v}$ **at top centerline of tank**

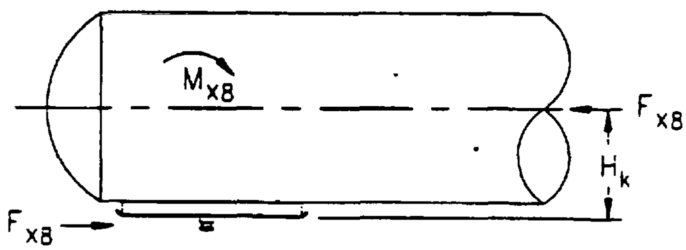
7.6 Determination of S_{x7}, the tensile stress generated by the axial load resulting from a longitudinal decelerative force.



$S_{x7} = \frac{F_{x7}}{a}$ at top, side, & bottom centerline of tank

where: $F_{x7} = 0.70\ (F_k + W_k)$

7.7 Determination of S_{x8} stress generated by the bending moment resulting from a longitudinal decelerative force.



$S_{x8} = \dfrac{\pm M_8}{Z_v}$ at top centerline of tank

$S_{x8} = 0$ at side centerline of tank

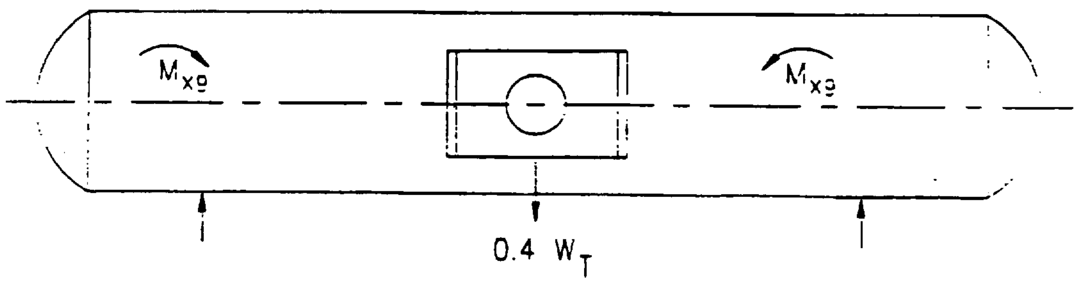
$S_{x8} = \dfrac{-M_{x8}}{Z_v}$ at bottom centerline of tank

where $M_{x8} = F_{x8}\ H_k$

Note: This equation gives the maximum value of M_{x8}. Upon analysis by a Design Certifying Engineer, M_{x8} may be found to decrease over the length of a cargo tank motor vehicle depending upon support assumptions.

but $F_{x8} = F_{x7} = 0.70\ (F_k + W_k)$

7.8 Determination of S_{x9} stress generated by a bending moment resulting from a lateral accelerative force.

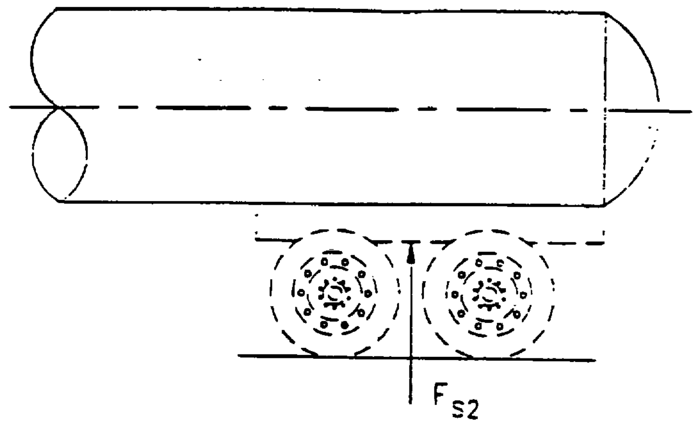


$$S_{x9} = \frac{M_{x9}}{Z_v} = \frac{(0.4)\ M_{max}}{Z_v}$$ at side centerline of tank

$S_{x9} = 0$ at top and bottom centerline of tank

$$S_{x9} = \frac{(0.4)\ M_{max}}{Z_v}$$ at side centerline of tank

7.9 Determination of S_{s2} vertical shear stress generated by an upward accelerative force.



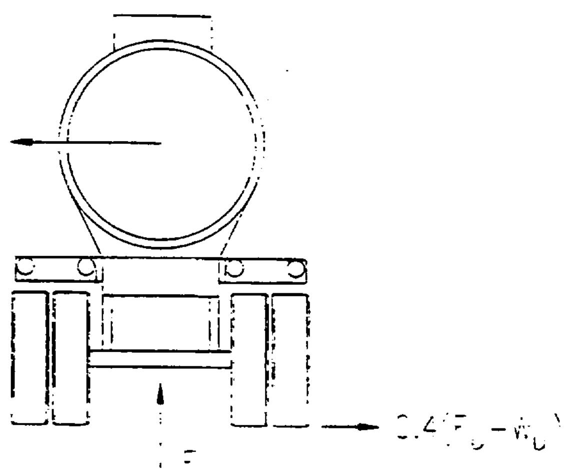
$S_{s2} = \frac{F_{s2}}{a/2}$ at side centerline of tank

$S_{s2} = 0$ at top and bottom centerline of tank

where: $F_{s2} = 0.70\ F_u$

Refer to shear diagram on page 17 for a more accurate calculation of shear force along the length of the trailer.

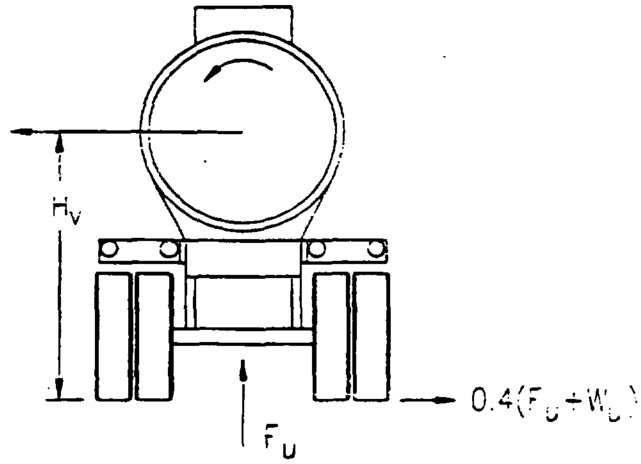
7.10 Determination of S_{s3} lateral shear stress generated by a lateral accelerative force.



$S_{s3} = \frac{0.4\ (F_u + W_u)}{a/2}$ at top and bottom centerline of tank

$S_{s3} = 0$ at side centerline of tank

7.11 Determination of S_{s4} torsional shear stress generated by a lateral accelerative force.



$$S_{s4} = \frac{T\,R_o}{J}$$ at top, side, & bottom centerline of tank

where

$T = 0.4\ (F_u + W_u)\ H_v$

8.0 Determination of the Effective Stress(s) at any Point Resulting from the Combination of Static and Dynamic Loadings that Occur at the Same Time:

8.1 Regulatory requirements.

8.1.1 According to 49 CFR 178.345-3(a)(1), the maximum calculated design stress at any point in the tank wall may not exceed the maximum allowable stress value prescribed in Section VIII of the ASME Code, or 25 percent of the tensile strength of the material used at design conditions."

Note: 49 CFR 171.7(a), *Matter incorporated by reference-* recognizes only the 1992 Edition, A93 Addenda of the ASME Code and does not allow current editions of the ASME Code to be incorporated.

According to 49 CFR 178.345-3(a)(2), "The relevant physical properties of the materials used in each cargo tank may be established either by a certified test report from the material manufacturer or by testing in conformance with a recognized national standard. In either case, the ultimate tensile strength of the material used in the design may not exceed 120 percent of the minimum ultimate tensile strength specified in either the ASME Code or the ASTM standard to which the material is manufactured."

CAUTION: Published minimum tensile strengths and certified test reports are normally specified at ambient temperatures. Aluminum especially, and stainless and carbon steel to a lesser degree, all have degrading tensile strengths with increased temperature. The tensile strength used shall be adjusted downward to reflect higher temperature ratings. Interpolation of the tensile strength used with respect to the ASME published allowable tensile strengths at the design temperature is one accepted method.

The Design Certifying Engineer should consider joint efficiencies and heat affected zones when determining the allowable strength. When applicable, the ASME Code, physical tests, or other acceptable means may be used to determine joint efficiencies.

8.1.2 According to 49 CFR 178.345-3(c), "Shell stresses resulting from static or dynamic loadings, or a combination thereof, are not uniform throughout the cargo tank motor vehicle. The effective stress (the maximum principal stress at any point) must be determined by the following formula.

$$S = 0.5\,(S_y + S_x) \pm [0.25\,(S_y - S_x)^2 + S_s^2]^{0.5}$$

Extreme Dynamic Loadings

8.2 Case A; Static Stress and Stress Due to Longitudinal Deceleration Created by Trailer Braking

For this combination, the effective stress in the cargo tank shell, S_A, is calculated based on the following stresses occurring simultaneously:

Circumferential stress due to pressure, S_y,

Longitudinal tensile (or compressive) stress due to pressure, and (static bending stress due to) the weight of the lading and the cargo tank, S_{x1},

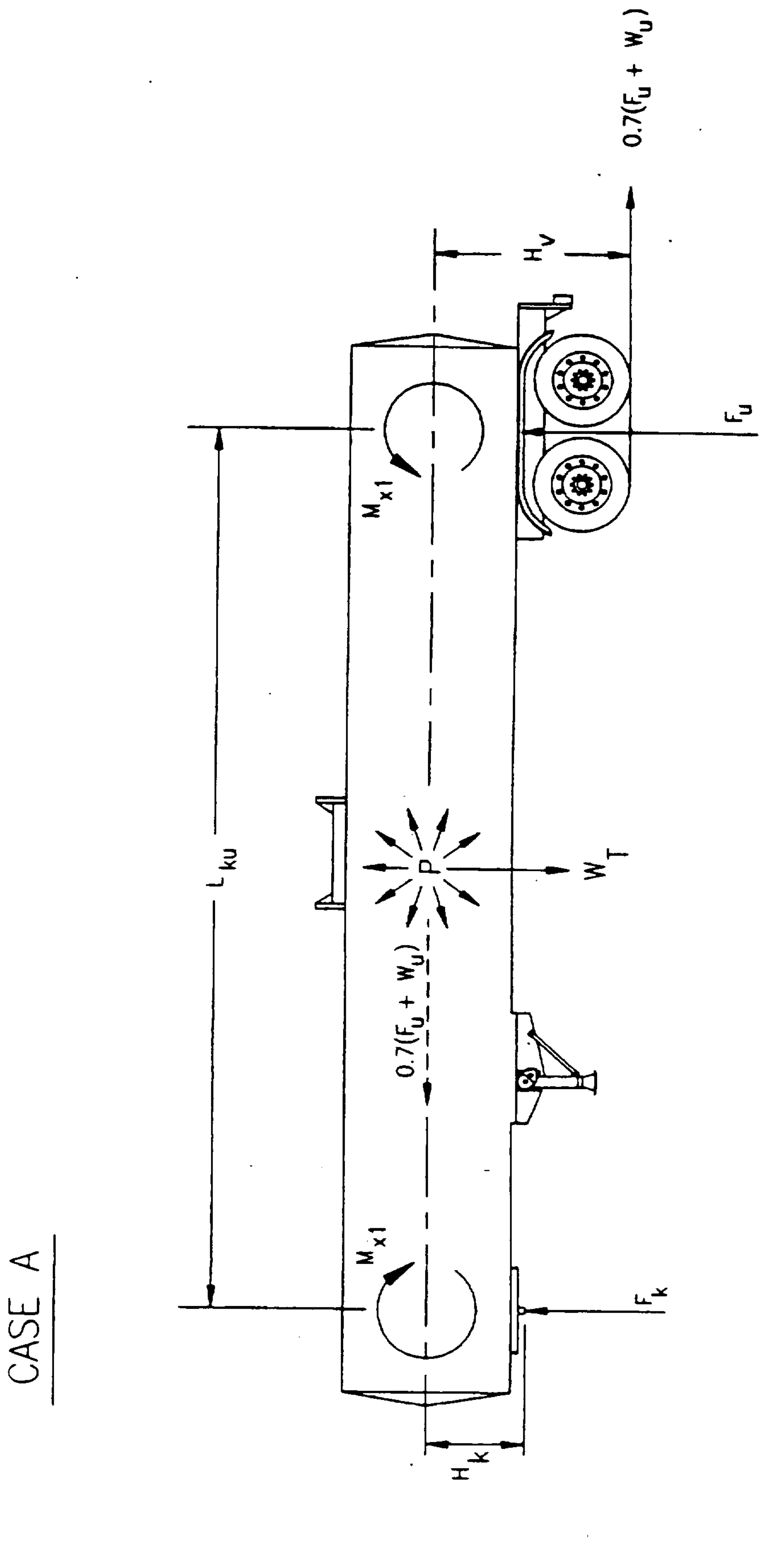
Longitudinal tensile (axial) stress due to longitudinal deceleration created by trailer braking, S_{x2},

Longitudinal tensile (or compressive) stress due to bending resulting from longitudinal deceleration created by trailer braking, S_{x3}, and

Static shear stress, S_{s1}.

$$S_A = 0.5(S_y+S_{xA})\pm[0.25(S_y-S_{xA})^2+S_{sA}^2]^{0.5},$$

where $S_{xA} = S_{x1}+S_{x2}+S_{x3}$, and
$S_{sA} = S_{s1}$.


CASE A
0.7(F_u + W_u)
H_v
F_u
M_x1
L_ku
P
W_T
0.7(F_u + W_u)
M_x1
F_k
H_k

8.3 Case AA; Static Stress and Stress Due to Longitudinal Deceleration Created by Tractor Braking

For this combination, the effective stress, S_{AA}, is calculated based on the following stresses occurring simultaneously:

Circumferential stress due to pressure, S_y,

Longitudinal tensile (or compressive) stress due to internal pressure, and (static bending stress due to) the weight of the lading and the cargo tank, S_{x1}

Longitudinal compressive (axial) stress due to longitudinal deceleration created by tractor braking, S_{x7},

Longitudinal tensile (or compressive) stress due to bending resulting from longitudinal deceleration created by tractor braking, S_{x8}, and

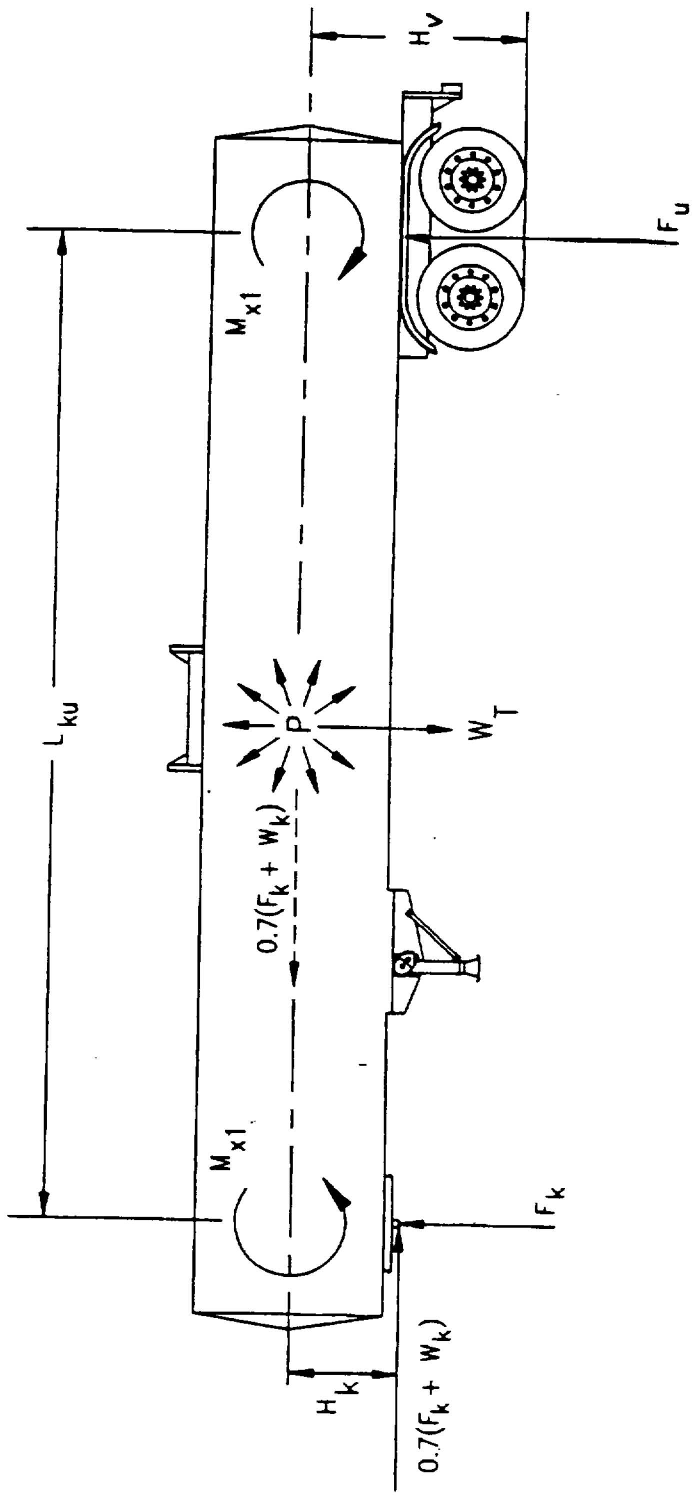
Static shear stress, S_{s1}.

$$S_{AA} = 0.5(S_y+S_{xAA})\pm[0.25(S_y-S_{xAA})^2+S_{sAA}^2]^{0.5},$$

where $S_{xAA} = S_{x1}+S_{x7}+S_{x8}$, and

$S_{sA} = S_{s1}$.

CASE AA



8.4 Case B; Static Stress and Stress Due to Longitudinal Acceleration

For this combination, the effective stress in the cargo tank shell, S_B, is calculated based on the following stresses occurring simultaneously:

Circumferential stress due to pressure, S_y,

Longitudinal tensile (or compressive) stress due to internal pressure, and (static bending stress due to) the weight of the lading and the cargo tank, S_{x1},

Longitudinal tensile (axial) stress due to longitudinal acceleration, S_{x4},

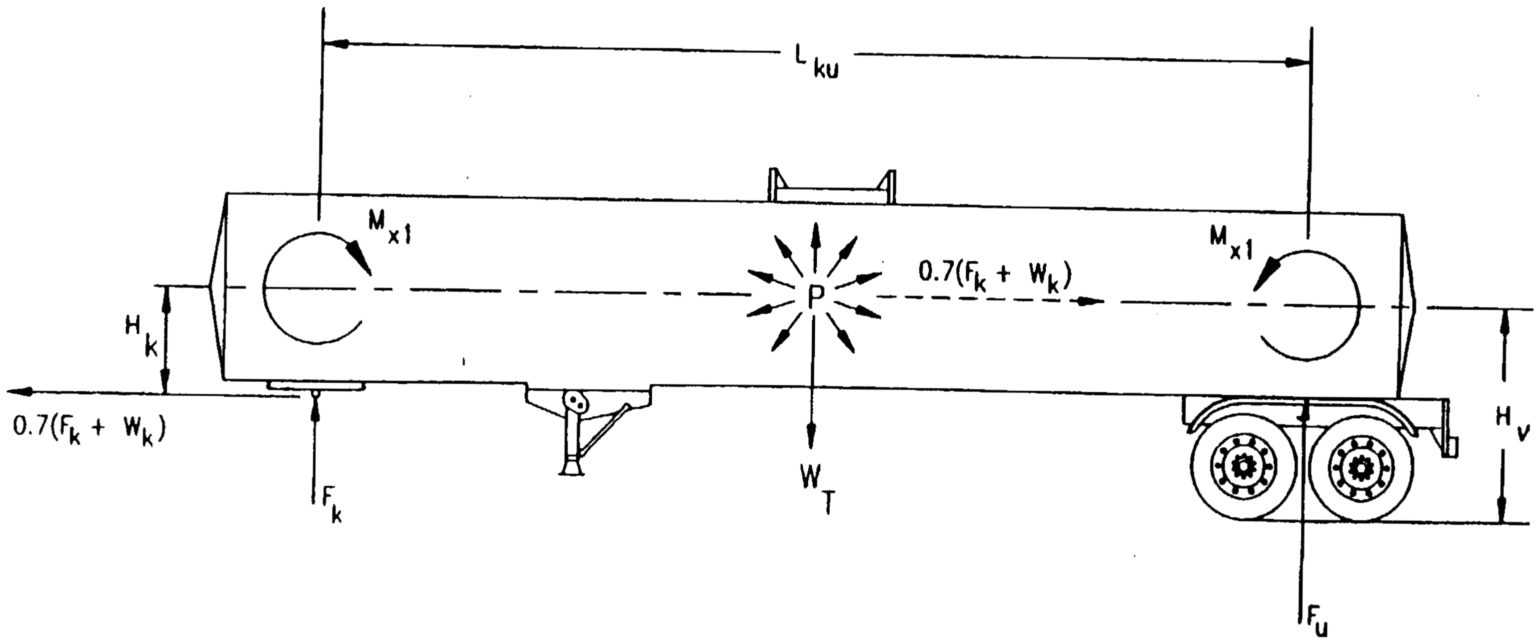
Longitudinal tensile (or compressive) stress due to bending resulting from longitudinal acceleration, S_{x5}, and

Static shear stress, S_{s1}.

$$S_B = 0.5(S_y+S_{xB})\pm[0.25(S_y-S_{xB})^2+S_{sB}^2]^{0.5},$$

where $S_{xB} = S_{x1}+S_{x4}+S_{x5}$, and
$S_{sB} = S_{s1}$.

CASE B



8.5 Case C; Static Stress and Stress Due to Vertical Acceleration

For this combination, the effective stress in the cargo tank shell, S_C, is calculated based on the following stresses occurring simultaneously:

Circumferential stress due to pressure, S_y,

Longitudinal tensile (or compressive) stress due to internal pressure, and (static bending stress due to) the weight of the lading and the cargo tank, S_{x1},

Longitudinal tensile (or compressive) stress due to bending resulting from a vertical acceleration occurring at both the kingpin and the suspension, S_{x6},

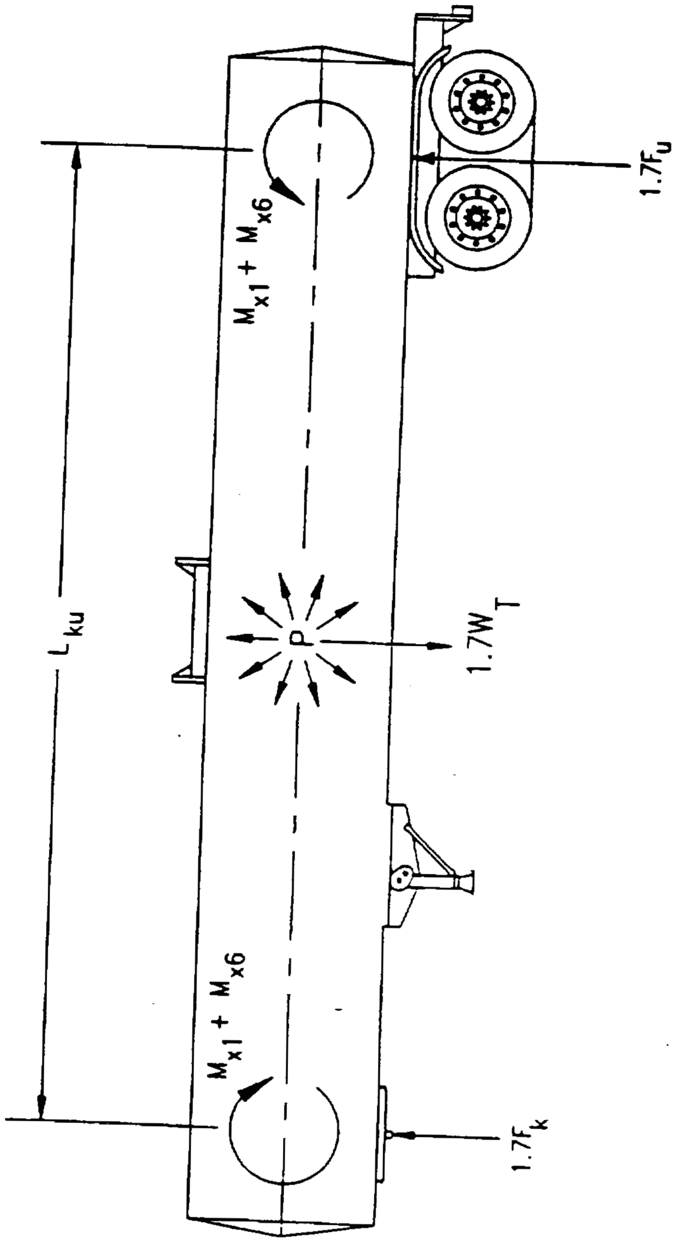
Static shear stress, S_{s1}, and

Shear stress generated by a vertical acceleration, S_{s2}.

$$S_C = 0.5(S_y+S_{xC})\pm[0.25(S_y-S_{xC})^2+S_{sC}^2]^{0.5},$$

where $S_{xC} = S_{x1}+S_{x6}$, and
$S_{sC} = S_{s1}+S_{s2}$

CASE C



8.6 Case D; Static Stress and Stress Due to Lateral Acceleration

For this combination, the effective stress in the cargo tank shell, S_D, is calculated based on the following stresses occurring simultaneously:

Circumferential stress due to pressure, S_y,

Longitudinal tensile (or compressive) stress due to internal pressure, and (static bending stress due to) the weight of the lading and the cargo tank, S_{x1},

Longitudinal tensile (or compressive) stress due to bending resulting from a lateral acceleration, S_{x9},

Static shear stress, S_{s1}.

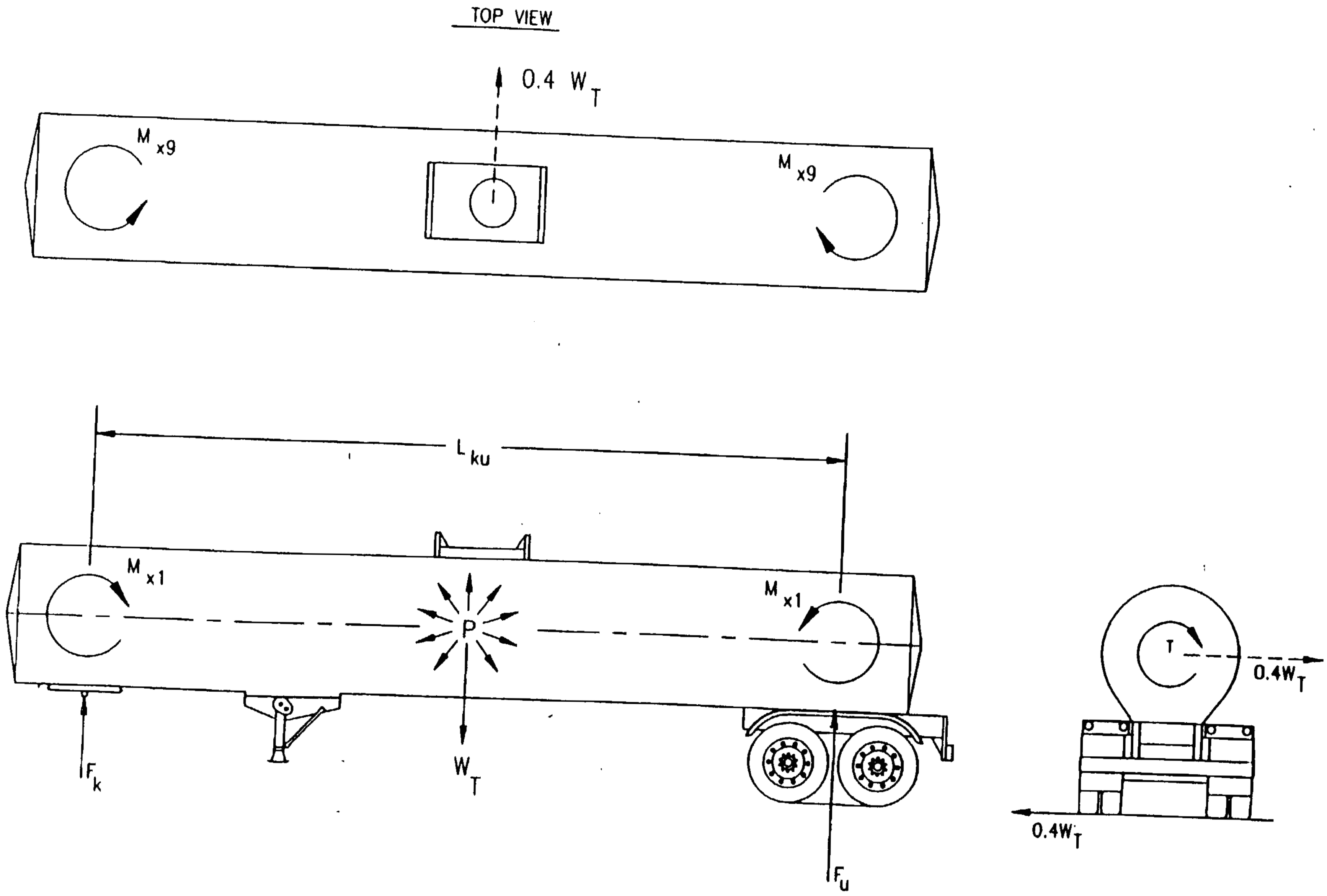
Shear stress due to a lateral acceleration, S_{s3}, and

Shear stress due to torsion generated by a lateral acceleration, S_4.

$$S_D = 0.5(S_y+S_{xD}\pm[0.25(S_y-S_{xD})^2+S_{sD}^2]^{0.5},$$

where $S_{xD} = S_{x1}+S_{x9}$, and
$S_{sD} = S_{s1}+S_3+S_4$.

CASE D



Normal Operating Loadings

8.7 Case E; Static Stress and Combined Stress Due to Longitudinal Deceleration Created by Trailer Braking, Vertical Acceleration, and Lateral Acceleration in Normal Operating Conditions

For this combination, the effective stress, S_E, is calculated based on the following stresses occurring simultaneously:

Circumferential stress due to pressure, S_y,

Longitudinal tensile (or compressive) stress due to internal pressure, and (static bending stress due to) the weight of the lading and the cargo tank, S_{x1}.

Longitudinal tensile (axial) stress due to longitudinal deceleration created by trailer braking, $(S_{x2}/2)$*,

Longitudinal tensile (or compressive) stress due to bending resulting from longitudinal deceleration created by trailer braking, $(S_{x3}/2)$*,

Longitudinal tensile (or compressive) stress due to bending resulting from a vertical acceleration at the suspension, $(S_{x6}/2)$*,

Longitudinal tensile (or compressive) stress due to bending resulting from a lateral acceleration, $S_{x9}/2$*,

Shear stress generated by static vertical acceleration, S_{s1},

Shear stress generated by dynamic vertical acceleration, $S_{s2}/2$*,

Shear stress due to a lateral acceleration, $S_{s3}/2$, and

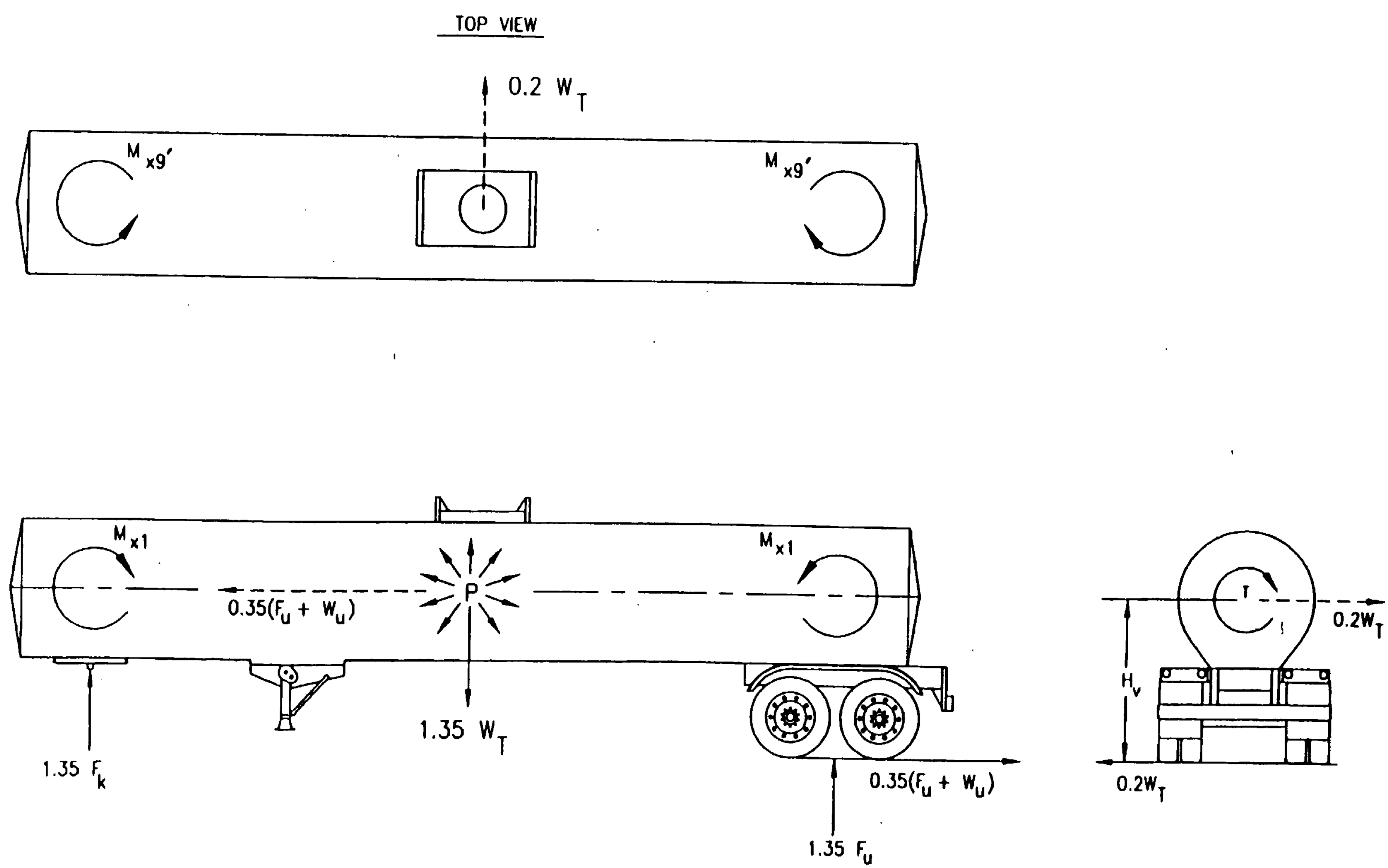
Shear stress due to torsion generated by a lateral acceleration, $S_{s4}/2$*.

$$S_E = 0.5(S_y+S_{xE})\pm[0.25(S_y-S_{xE})^2+S_{sE}^2]^{0.5},$$

where $S_{xE} = S_{x1}+(S_{x2}+S_{x3}+S_{x6}+S_{x9})/2$, and
$S_{sE} = S_{s1}+(S_{s2}+S_{s3}+S_{s4})/2$.

* The magnitudes of these accelerations are precisely one-half of those prescribed in 49 CFR 178.345-3; 0.35G vertical, 0.35G longitudinal, and 0.2G lateral, respectively.

CASE E


TOP VIEW
0.2 W_T
$M_{x9'}$
$M_{x9'}$
M_{x1}
M_{x1}
P
0.35(F_u + W_u)
1.35 W_T
1.35 F_k
1.35 F_u
0.35(F_u + W_u)
T
0.2W_T
H_v
0.2W_T

8.8 Case EE; Static Stress and Combined Stress Due to Longitudinal Deceleration Created by Tractor Braking, Vertical Acceleration, and Lateral Acceleration in Normal Operating Conditions.

For this combination, the effective stress, S_{EE}, is calculated based on the following stresses occurring simultaneously.

Circumferential stress due to pressure, S_y,

Longitudinal tensile (or compressive) stress due to internal pressure, and (static bending stress due to) the weight of the lading and the cargo tank, S_{x1},

Longitudinal tensile (or compressive) stress due to bending resulting from a vertical acceleration at the kingpin, $(S_{x6}/2)$*,

Longitudinal compressive (axial) stress due to longitudinal deceleration created by tractor braking, $(S_{x7}/2)$*,

Longitudinal tensile (or compressive) stress due to bending resulting from longitudinal deceleration created by tractor braking, $(S_{x8}/2)$*,

Longitudinal tensile (or compressive) stress due to bending resulting from a lateral acceleration, $S_{x9}/2$*,

Shear stress generated by static vertical acceleration, S_{s1},

Shear stress generated by dynamic vertical acceleration, $S_{s2}/2$*,

Shear stress due to a lateral acceleration, $S_{s3}/2$*, and

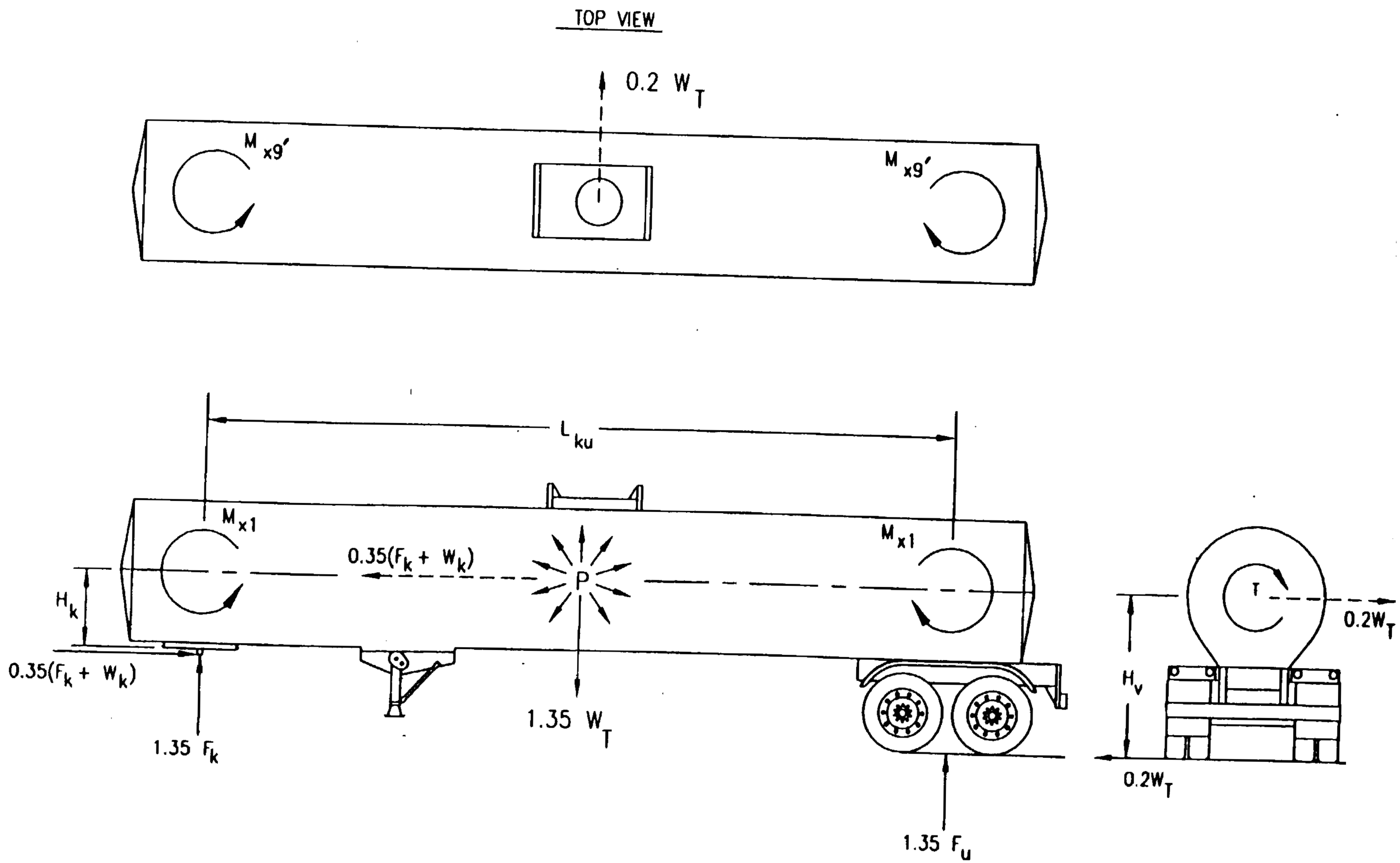
Shear stress due to torsion generated by a lateral acceleration, $S_{s4}/2$*.

$$S_{EE} = 0.5(S_y+S_{xEE})\pm[0.25(S_y-S_{xEE})^2+S_{sEE}{}^2]^{0.5},$$

where $S_{xEE} = S_{x1}+(S_{x6}+S_{x7}+S_{x8}+S_{x9})/2$, and
$S_{sEE} = S_{s1}+(S_{s2}+S_{s3}+S_{s4})/2$.

* The magnitudes of these accelerations are precisely one-half of those prescribed in 49 CFR 178.345-3; 0.35G vertical, 0.35G longitudinal and 0.2G lateral.

CASE EE



TOP VIEW
0.2 W_T
$M_{x9'}$
$M_{x9'}$
L_{ku}
M_{x1}
M_{x1}
0.35(F_k + W_k)
P
H_k
0.35(F_k + W_k)
1.35 F_k
1.35 W_T
1.35 F_u
T
0.2W_T
H_v
0.2W_T

8.9 Case F; Static Stress and Combined Stress Due to Longitudinal Acceleration, Vertical Acceleration, and Lateral Acceleration in Normal Operating Conditions.

For this combination, the effective stress, S_F, is calculated based on the following stresses occurring simultaneously.

Circumferential stress due to pressure, S_y.

Longitudinal tensile (or compressive) stress due to internal pressure, and (static bending stress due to) the weight of the lading and the cargo tank, S_{x1},

Longitudinal tensile (axial) stress due to longitudinal acceleration, $S_{x4}/2$*,

Longitudinal tensile (or compressive) stress due to bending resulting from longitudinal acceleration, $S_{x5}/2$*,

Longitudinal tensile (or compressive) stress due to bending resulting from a vertical acceleration at the kingpin, $(S_{x6}/2)$*,

Longitudinal tensile (or compressive) stress due to bending resulting from a lateral acceleration, $S_{x9}/2$*,

Shear stress generated by static vertical acceleration, S_{s1},

Shear stress generated by dynamic vertical acceleration, $S_{s2}/2$*,

Shear stress due to a lateral acceleration, $S_{s3}/2$*, and

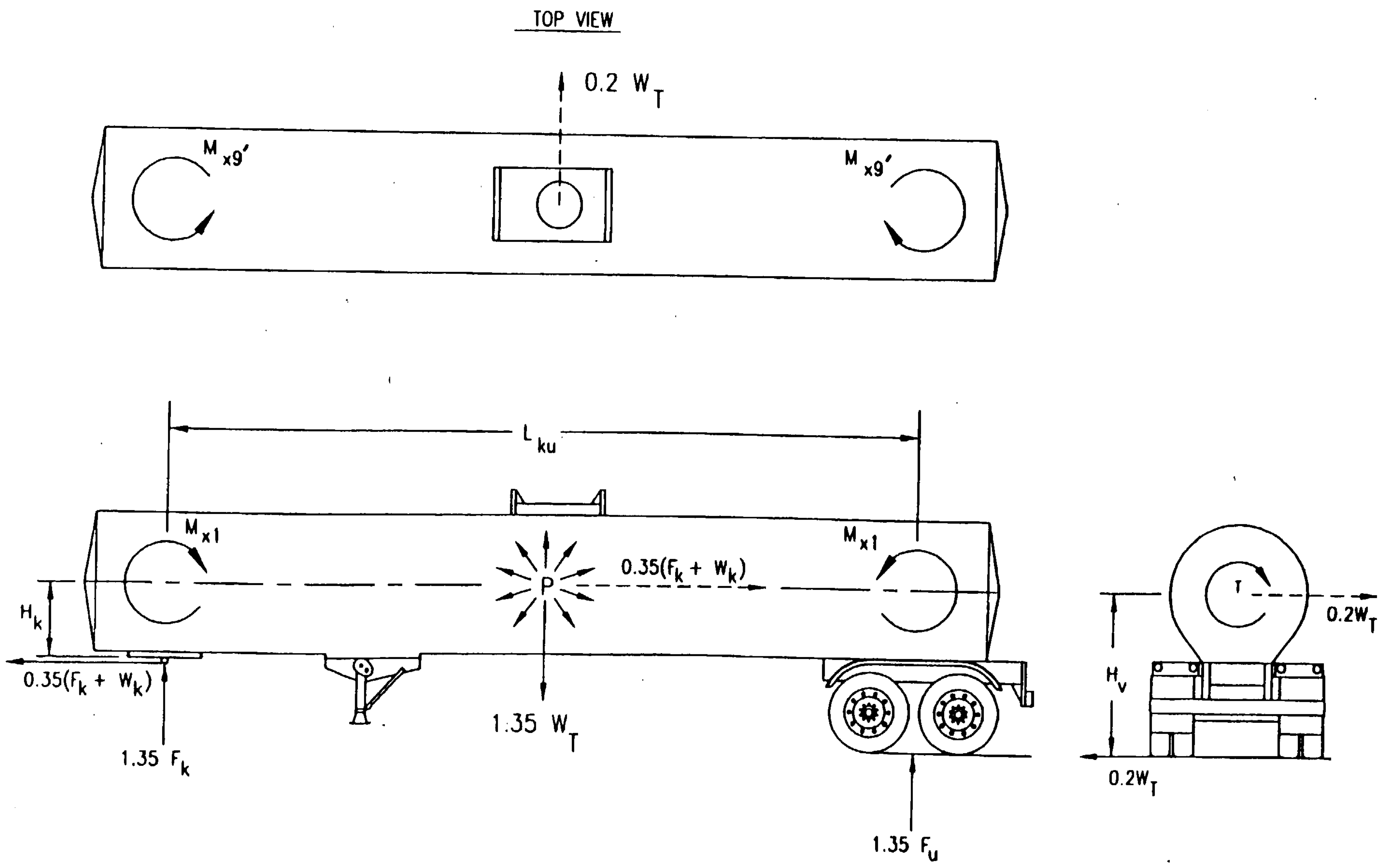
Shear stress due to torsion generated by a lateral acceleration, $S_{s4}/2$*.

$$S_F = 0.5(S_y+S_{xF})\pm[0.25(S_y-S_{xF})^2+S_{sF}^2]^{0.5},$$

where $S_{xF} = S_{x1}+(S_{x4}+S_{x5}+S_{x6}+S_{x9})/2$, and
$S_{sF} = S_{s1}+(S_{s2}+S_{s3}+S_{s4})/2$.

* The magnitudes of these accelerations are precisely one-half of those prescribed in 49 CFR 178.345-3, 0.35 vertical, 0.35G longitudinal, and 0.2G lateral.

CASE F


TOP VIEW
0.2 W_T
$M_{x9'}$
$M_{x9'}$
L_{ku}
M_{x1}
M_{x1}
P
0.35(F_k + W_k)
H_k
0.35(F_k + W_k)
1.35 W_T
1.35 F_k
1.35 F_u
T
0.2W_T
H_v
0.2W_T

9.0 Determination of Head and Shell Stresses Due to Accident Induced Loads:

9.1 Regulatory requirements.

9.1.1 The tensile stress determined as a result of the accident induced deceleration is not combined with the static and dynamic stresses except as noted in 9.1.2 since 178.345-3(a)(3) states that - "The maximum design stress at any point in the cargo tank must be calculated separately for the loading conditions described in paragraphs (b) [static loads], (c) [static & dynamic loads].

9.1.2 49 CFR 178.345-8(e) states that -

(e) Longitudinal deceleration protection.
In order to account for stresses due to longitudinal impact in an accident, the tank shell and heads must be able to withstand the load resulting from the design pressure in combination with the dynamic pressure resulting from a longitudinal deceleration of 2 "g". For this loading condition, the allowable stress value used may not exceed the ultimate strength of the material of construction using a safety factor of 1.3. Performance testing, analytical methods, or a combination thereof, may be used to prove this capability provided the methods are accurate and verifiable. For cargo tanks with internal baffles, the decelerative force may be reduced by 0.25 "g" for each baffle assembly, but in no case may the total reduction in decelerative force exceed 1.0 "g".

To allow for physical testing, the safety factor of 1.3 may be applied to the 2"g" load and use the ultimate tensile strength as the allowable stress.

9.2 Determination of the shell stress due to liquid product dynamic pressure resulting from an accident induced longitudinal deceleration in conjunction with static pressures.

$$P_L = P_d + P_h + P_m$$

$$P = (.4336) \left[\frac{L_o}{12}\right] \left[\frac{W_L}{(8.3)\ V}\right] (2\text{-}0.25b)$$

where .4336 is conversion from feet of water to psi.

$L_o/12$ is the height of the longitudinal static head.

$W_L/(8.3)V$ is the specific gravity of the lading.

2 - 0.25b is the "2g" deceleration minus "0.25g" credit for each baffle where $b \leq 4$

$$P_h = (.4336) \left[\frac{D_i}{12}\right] \left[\frac{W_L}{(8.3)\ V}\right]$$

where $\frac{D_i}{12}$ is the height of the vertical static head.

P_m = MAWP (this is for the highest pressure as the pressure relief could be set lower).

Therefore,

$$P_L = \frac{(.4336)\ (W_L)}{(8.3)\ V\ (12)} [(2\text{-}0.25b)\ L_o + D_i] + P_m$$

$$= (4.3534 \times 10^{-3})\ \frac{(W_L)}{V}\ ((2\text{-}0.25b)\ L_o + D_i) + P_m$$

In the case of longitudinal stress,

$$S_{dx} = \frac{P_L\ R_i}{2\ t_s}$$

In the case of circumferential stress,

$$S_{dy} = \frac{P_L\ R_i}{t_s}$$

Since S_{dy} is greater than S_{dx}.
let $S_d = S_{dy}$

9.3 Determination of the torispherical head stress due to liquid product dynamic pressure resulting from an accident induced longitudinal deceleration in conjunction with static pressures.

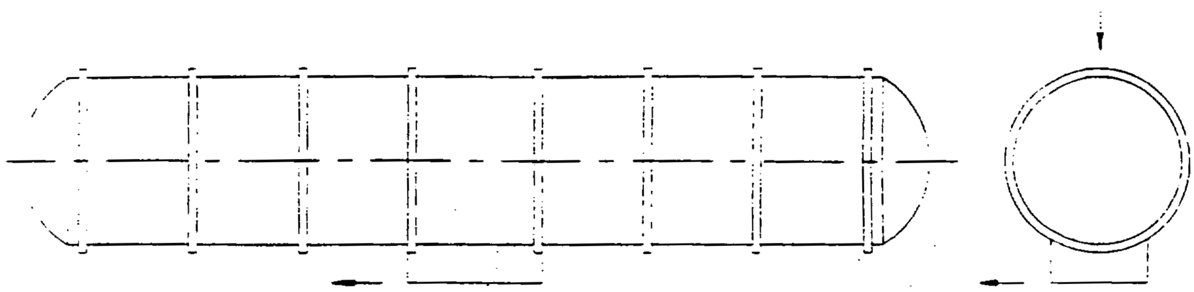
$$S_d = \frac{P_L\ R_h\ Q}{2\ t_h}$$

$$\text{where } Q = 1/4 \left[3 + \left[\frac{R_h}{R_k}\right]^{1/2}\right]$$

The torispherical head stress for the case in which the knuckle radius is 6% of the inside crown radius and the inside crown radius equals the outside diameter of the shirt, shall be determined by

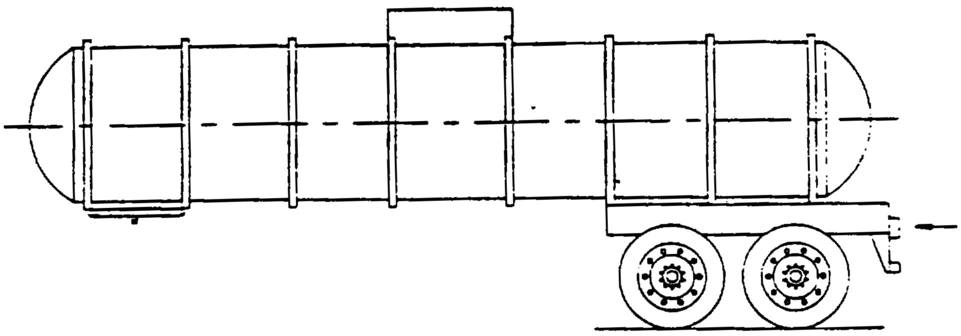
$$S_{ds} = \frac{0.885 P_L\ R_h}{t_h}$$

9.4 Determination of the stresses in the shell due to an accident induced overturn.



The calculation of the stresses in the cargo tank due to accident damage protection loads shall be considered, but are not within the scope of this Recommended Practice.

9.5 Determination of the stresses in the shell due to an accident induced rear end impact.



The calculation of the stresses in the cargo tank due to accident damage protection loads shall be considered, but are not within the scope of this Recommended Practice.

10. Verification that the Head and Shell Thickness is Equal to or Greater than that Specified by DOT.

10.1 49 CFR 178.345-3(e) states that --

> "In no case may the minimum thickness of the cargo tank wall be less than that prescribed in 178.346-2, 178.347-2, or 178.348-2, as applicable."

TTMA/DWV/mm